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A ROOM FIRE MODEL IN VIEW OF PREDICTING FIRE SPREAD BY EXTERNAL FLAMES

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ABSTRACT

A model for predicting room fire in fully developed stage is developed for the purpose of estimating the heat release due to the combustion of the excess fuels involved in the external flames ejecting from the windows of fire rooms. In this model, the burning rate is predicted as a function of the predicted heat transfer to the fuels. The model does not need to introduce adjusting factors, such as the "complete combustion rate (=0.6)" by Kawagoe et al., to get reasonably good prediction of room fire temperature. The predictions of the temperatures and the burning rates are compared with the results of preceding experiments to demonstrate fair agreement.

1. INTRODUCTION

A significant number of models have been developed to date for the prediction of room fire behavior, for such purposes as assessing the fire impact to structural members or the risk of the fire spread to adjacent spaces. In many of them, however, the burning rate is not predicted but given independently from the heat transfer conditions within the room.

A typical example can be seen in the model by Kawagoe & Sekine, which is well known and widely used in the fire resistance community in Japan[1]. This model numerically solves the energy conservation equation given as follows:

$$Q_C = Q_W + Q_R + Q_S \qquad \cdots (1)$$

where Q_C, Q_W, Q_R and Q_S are the rates of the heat release by the combustion, the heat transfer to the boundary walls, the radiative heat loss from the opening and the heat convected out by the opening flow, respectively. In this model the heat release rate Q_C is given by the product of the heat of combustion $\triangle H_C$ and the mass burning rate M, for the latter of which the famous relationship:

$$M \approx 0.1A\sqrt{H} \qquad \cdots (2)$$

empirically derived by Kawagoe et al. is invoked. However, the prediction by this method yields unreasonably high temperature, hence so-called "complete combustion rate" with fixed value of 0.6 has been introduced to fill up the gap with the experimentally measured temperatures.

It is considered that this discrepancy is caused by that the model by Kawagoe et al. does not take into account the portion of the fuel volatile that cannot burn within the room due to the shortness of

available air so is ejected out to burn as external flames. This portion will not contribute to the elevation of the temperature of the room but will affect the temperature of the opening flames and plumes.

Although the distance between the openings on adjacent floors are often controlled by building codes to prevent upstairs fire spread, it is not clear if the provisions is sufficiently rational. The hazard of upstairs fire spread must be a complex function of opening dimension, room size, fire load etc. as well as the distance between the openings. In this study, a room fire model incorporating the prediction of mass burning rate is developed so that the heat release rate due to the combustion of the excess fuels in opening flames can be estimated under different conditions of fire room and fuels.

The intention is to estimate the heat release rate Q_{ex} due to the combustion of excess fuel by

$$Q_{ex} = \Delta H_{fuel} M - \Delta H_{air} R_{OF} \quad \cdots (3)$$

using the mass burning rate M predicted by the model.

2. ROOM FIRE MODEL

This model is a single room fire model which assumes that the physical properties is uniform within the fire room, like most of the other models for similar purpose. The concept of the model is illustrated in Figure 1.

2.1 Conservation and State of Room Gases

Considering the conservation of the mass, the energy and the species for the gases in the fire room, the following equations can be obtained.

(1) Mass Conservation

$$\frac{d}{dt}(\rho V) = R_{OF} - R_{FO} + M \quad \cdots (4)$$

(2) Energy Conservation

$$\frac{d}{dt}(c_P \rho T V) = Q_C - Q_W - Q_R - Q_{fuel} + C_P R_{OF} T_O - C_P R_{FO} T + C_P T_P M \quad \cdots (5)$$

Note that the rate of the heat transfer to fuels Q_{fuel} is included in Eqn.(5) separately from the heat transfer to room boundary walls.

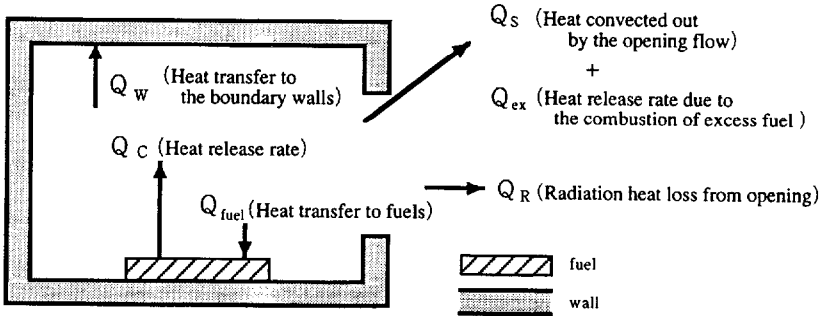

Figure 1. Concept of model

(3) Species Conservation

$$\frac{d}{dt}(\rho V Y_l) = R_{OF} Y_{l,O} - R_{FO} Y_l + M Y_{l,fuel} + \Gamma_l \qquad \cdots (6)$$

2.2 Heat Transfer Models

(1) Heat transfer to the boundary walls

The room boundary walls are assumed to be uniform in thermal properties and in temperature, and the rate of heat transfer Q_W is calculated using an ordinary equation as:

$$\frac{Q_W}{A_W} = \varepsilon_W \varepsilon_G \sigma\left(T^4 - T_W^4\right) + h\left(T - T_W\right) \qquad \cdots (7\text{-}1)$$

The surface temperature of the wall T_W is calculated by solving the one dimensional thermal conduction equation for the wall.

(2) Heat transfer to fuels

The rate of the heat transfer to the fuels is calculated by the same equation as Eqn.(7-2) except that fuel gasification temperature T_P is used instead of wall surface temperature T_W.

$$\frac{Q_{fuel}}{A_{fuel}} = \varepsilon_{fuel} \varepsilon_G \sigma\left(T^4 - T_P^4\right) + h\left(T - T_P\right) \qquad \cdots (7\text{-}2)$$

However, the rate of the heat transfer for wood fuels is assumed to be 1/4 of Q_{fuel} thus calculated by Eqn.(7-2) based on the results of the preceding experiments[2].

(3) Radiation heat loss from opening

The heat loss due to the radiation through the opening of a room Q_R is simply given as:

$$Q_R = \varepsilon_G \sigma\left(T^4 - T_O^4\right) A \qquad \cdots (8)$$

(4) Convective heat transfer coefficient

No established method to estimate the convective heat transfer coefficient h included in Eqns.(7-1) and (7-2) exists to date, but the following formula is used for its rough estimation[3]

$$h = \begin{cases} 5 \times 10^{-3} & (T^* \le 300K) \\ (0.02T^* - 1) \times 10^{-3} & (300K < T^* < 800K) \\ 15 \times 10^{-3} & (800K \le T^*) \end{cases} \qquad \cdots (9)$$

where T * is defined as

$$T^* = \begin{cases} (T + T_W)/2 & (for\ wall) \\ (T + T_P)/2 & (for\ fuel) \end{cases} \qquad \cdots (10)$$

(5) Emissivity of room fire gases

The fire gases in a real scale room can be regarded as black, but in this model the emissivity of the fire gases is calculated for the convenience of making comparison with the results of the preceding experiments, for which a somewhat small scale room was used.

(a) Emissivity due to smoke particles

The emissivity of the fire gases due to smoke particles in the fire room is calculated using the equation given by Quintiere[4] as follows:

$$\varepsilon_G = 1 - e^{-k_G L_G} \quad \cdots (11)$$

where the effective absorption coefficient is given as

$$k_G = 1.3(\gamma + 1)\frac{M}{M + R_{OF}} \quad \cdots (12)$$

(b) Emissivity due to CO_2 and H_2O gases

The emissivity of the fire gases due to CO_2 and H_2O is calculated simply as the sum of the emissivity of each gas, namely:

$$\varepsilon_G = \varepsilon_{CO_2} + \varepsilon_{H_2O} \quad \cdots (13)$$

where the emissivities of CO_2 and H_2O are calculated by Schack's equations[5] given as

$$\varepsilon_{CO_2} = 0.7(P_{CO_2} \cdot L_G)^{0.5} / (T/100)^{0.5} \quad \text{and} \quad \varepsilon_{CO_2} = 7.0(P_{H_2O} \cdot L_G)^{0.8} / (T/100) \quad \cdots (14)$$

2.3 Mass Burning Rate

In this model, an initial fire source with given mass burning rate is employed to initiate the fire in the room, and the mass burning rate afterwards is calculated as a function of the heat transfer conditions in the room.

(1) Mass burning rate of initial fire source

The mass burning rate of the initial fire source M_1 is set as:

$$M_1 = m''_{free} A_{fuel} \quad \cdots (15)$$

In the computation, the burning rate is increased linearly from 0 to this value in one minute, since this model focuses on the behavior of vigorous stage of fire so does not concern the behavior in the initial stage.

(2) Mass burning rate in fuel controlled stage

It is assumed that the burning rate in fuel controlled stage M_2 is affected by the heat from the room gases as well as the heat from the flames formed above the fuels. When the temperature of the room is higher than the gasification temperature of the fuel, is calculated by

$$M_2 = m''_{free} A_{fuel} + Q_{fuel} / L_V \quad \cdots (16)$$

It is doubtful that the heat from the fire room gases can reach the fuel surface penetrating the flames above the fuels. However, it will reasonable to consider that the temperature rise of the room

reduces the net heat loss of the flames to the surrounding and thus indirectly increases the heat transfer to the fuel.

(3) Mass burning rate in ventilation controlled stage

In ventilation controlled stage, the combustion heavily depends on the air inflow through openings and the flames no longer locate in the vicinity of the fuel but often whirl around in the room. The mass burning rate at such a stage M_3 is considered to be controlled mainly by the heat transfer from the room fire gases, hence is calculated by

$$M_3 = Q_{fuel} / L_V \qquad \cdots (17)$$

(4) Mass burning rate in transitional stage

Figure 2 shows the transition of mass burning rate that may take place within fire room. In Figure 2, transition from M_1 to M_2 (①) begins when the room temperature reaches to the fuel gasification temperature , and M_2 increases as the net incident heat flux to the fuel increase. For ventilation controlled stage it will be reasonable to assume that the mass burning rate is given by M_3. However, switching from M_2 to M_3 at point② causes gaps by $m''_{free} A_{fuel}$. In this model, Eqn.(18) is used to manipulate the mass burning rate in the transitional stage from ② to ③, namely.

$$M_{2\to 3} = \frac{\Delta H_{air} R_{OF}}{\Delta H_{fuel}} \qquad \cdots (18)$$

This manipulation may be partly justified. Although the transition from fuel controlled stage to ventilation controlled stage takes place at point in terms of the heat release rate , the transition takes place somewhat more gradually in terms of mass burning rate. Therefore, the stage from ② to ③ is the stage where the behavior of the mass burning rate is changing gradually from that of the fuel controlled stage to that of the ventilation controlled stage . In any case, this is a relatively short period of time.

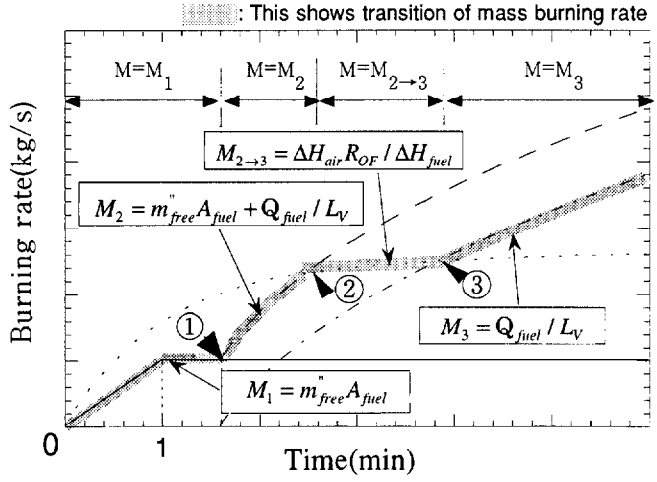

Figure 2. Transition of mass burning rate

2.4 Heat Release Rate within Fire Room

Since sufficient air is available to burn all the fuel volatile produced in the fire room in fuel controlled stage, the heat release rate is governed by the mass burning rate. On the other hand, since much more fuel volatile than can be burned by the air supplied into the room by the ventilation is produced in ventilation controlled stage, the heat release rate within the fire room is governed by

the air supply rate, hence

$$Q_C = \begin{cases} \Delta H_{fuel} M & (for\ fuel\ control) \\ \Delta H_{air} R_{OF} & (for\ ventilation\ control) \end{cases} \quad \cdots (19)$$

Note that the heat of combustion per unit mass of air consumed $\triangle H_{air}$ is 3MJ/kg in sufficient accuracy regardless the fuel type.

2.5 Species Generation Rate

Since it is known that the heat of combustion per unit mass of air consumed is virtually constant for many fuels encountered in fire, the generation (consumption) rate of oxygen can be expressed as

$$\Gamma_{O_2} = -\frac{Q_C}{\Delta H_{O_2}} \quad \cdots (20)$$

hence, when the chemical equation of the combustion of the fuel can be written as

$$fuel + \nu'_{O_2} O_2 \rightarrow \nu''_{CO_2} CO_2 + \nu''_{H_2O} H_2O \quad \cdots (21)$$

the generation rates of CO_2 and H_2O are given as follows:

$$\Gamma_{CO_2} = \frac{Q_C}{\Delta H_{O_2}} \times \frac{44}{32} \times \frac{\nu''_{CO_2}}{\nu'_{O_2}} \quad \text{and} \quad \Gamma_{H_2O} = \frac{Q_C}{\Delta H_{O_2}} \times \frac{18}{32} \times \frac{\nu''_{H_2O}}{\nu'_{O_2}} \quad \cdots (22)$$

3. Zone Equations

From Eqns.(4) - (6), the ordinary differential equations for the temperature and the species concentration of the fire room, and the algebraic equation for governing the ventilation of the room are obtained as follows:

(1) Room temperature

$$\frac{dT}{dt} = \frac{T}{C_P \rho_O T_O V}\left[\left(Q_C - Q_W - Q_R - Q_{fuel}\right) - C_P\left\{R_{OF}\left(T - T_O\right) + M\left(T - T_P\right)\right\}\right] \quad \cdots (23)$$

(2) Species mass concentration

$$\frac{dY_l}{dt} = \frac{T}{\rho_O T_O V}\left\{R_{OF}\left(Y_{l,O} - Y_l\right) + M\left(Y_{l,fuel} - Y_l\right) + \Gamma_l\right\} \quad \cdots (24)$$

In this model, the concentration of species l in fuel $Y_{l,fuel}$ is regarded as

$$Y_{l,fuel} = \begin{cases} 1 & (l = fuel) \\ 0 & (l = O_2, CO_2, H_2O) \end{cases} \quad \cdots (25)$$

(3) Ventilation

$$\frac{T_O}{T} R_{OF} - R_{FO} + \frac{T_P}{T} M + \frac{1}{C_P T}\left(Q_C - Q_W - Q_R - Q_{fuel}\right) = 0 \quad \cdots (26)$$

Here, Eqns.(23) and (24) are numerically integrated using Runge-Kutta method and Eqn.(26) is solved using Regula-Falsi method.

4. COMPARISON BETWEEN PREDICTIONS AND EXPERIMENTS

The predictions by the room fire model are compared with the results of the experiments[6] previously conducted to examine the validity of the model. In the experiments, methanol pools, PMMA panels, and wood panels and cribs with several different surface area were burned in the fire room with the dimensions of 1.1x1.8x1.1(m) under different opening conditions. And the room temperatures, the ceiling and the floor heat fluxes, and the mass burning rates were measured.

4.1 Conditions for Prediction and Experiments

The conditions of the calculations, which are essentially the same as the conditions of the preceding experiments, are shown in Table 1. Although the size of the model fire room is fixed for every case, some number of different conditions are tried for the opening. Methanol, PMMA and wood with different surface area are employed as the fuels.

4.2 Fire Room Temperature

Figure 3 shows a example of the transient temperatures predicted by the model and measured in the preceding experiments. In this example, the fuel is methanol with three different liquid surface areas and the value of the opening factor is $0.2m^{5/2}$. The temperatures in Figure 3(b) are the average of the temperature measured in the experiments by the 10 thermocouples arrayed vertically with the same spacing. It is considered that the fire with fuel A1 is completely fuel controlled , the fire with fuel A4 is completely ventilation controlled and the fire with fuel A2 is around the limit of

Table 1. Conditions of calculations[5][7]

	Unit	Values								
(Opening conditions)										
Width	m	0.4,0.6,0.8			0.2～0.9 (0.1m spacing)					
Height	m	0.3			1					
(Compartment conditions)										
Width	m	1.1								
Depth	m	1.8								
Height	m	1.1								
(Wall conditions)										
Thickness	m	0.1								
Thermal conductivity	kW/m/K	0.0002								
Thermal capacity	kJ/kg/K	1.1								
Density	kg/m^3	750								
(Fuel conditions)										
	(Type)	*A(1)*	*A(2)*	*A(4)*	*P(1)*	*P(2)*	*P(4)*	*W(1)*	*W(2)*	*W(C)*
Fuel		Methanol			PMMA			Wood		
Surface area	m^2	0.2	0.41	0.81	0.13	0.22	0.42	1.33	2.41	2.62
Weight()	kg	3.17			2.4			13	13	8
Latent heat of gasification	kJ/kg	1170			1630			1700		
Heat of combustion	kJ/kg	22300			25200			18400		
Stoichiometric air/fuel ratio		6.47			8.28			5.7		
Gasification temperature	K	338			543			623		
Outside temperature	K	293								

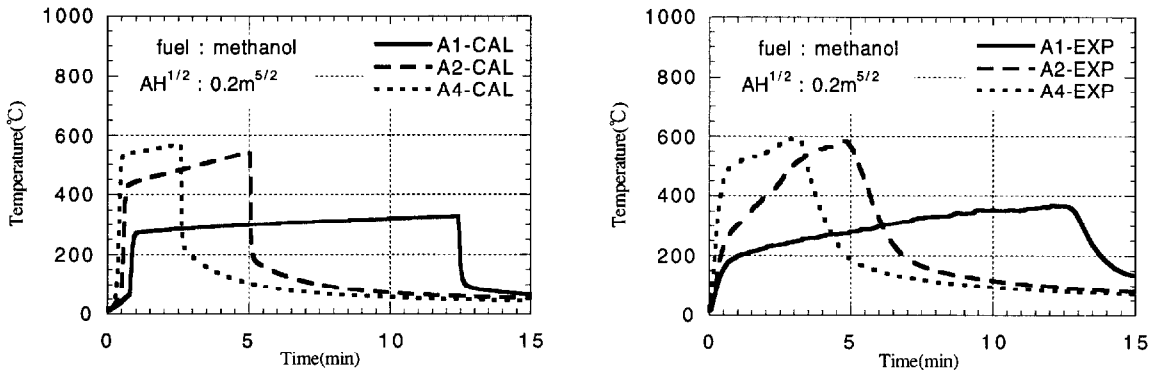

Figure 3. Temperature

the two regimes. From Figure 3, prediction seem to agree reasonably well with measurement, despite of liquid surface area.

Figure 4 compares the predicted and the measured temperature at vigorous burning stages. The temperatures of the predictions were taken from the values at the last moment of the fire duration, and the value of the experiments are the average during the vigorous burning stage. The open symbols indicate the cases where external flames were not observed in the experiment while the solid symbols indicate the cases where external flames were observed. Although the agreement between the predictions and measurements is only fair for methanol and PMMA, the agreement for wood, which is most common fuel in ordinary fires, seems satisfactory. The predicted temperatures generally tend to be a little higher than the measured temperatures. This may be partly due to the way the comparison is made: the latter is the average while the former is the highest.

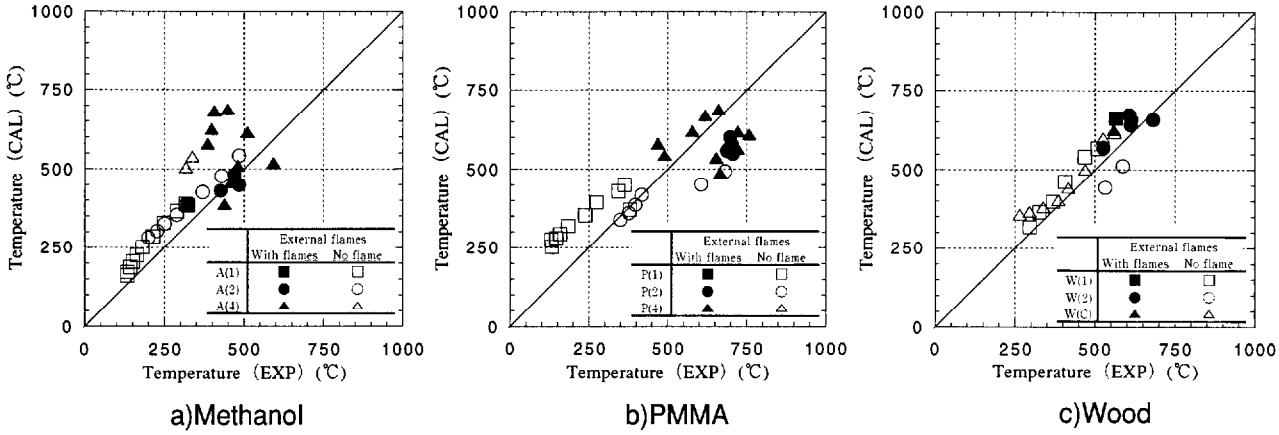

Figure 4. Comparison the predicted and the measured temperature

4.3 Mass burning rate

Figure 5 shows a example of the transient mass burning rates predicted by the model and measured in the preceding experiments. The calculation and experimental conditions of the example are the same as in the cases in Figure 3. In Figure 5, in case the liquid surface area is small, prediction seem to agree well with measurement. In case the liquid surface area is large, the prediction tend to be a little higher than the measurement.

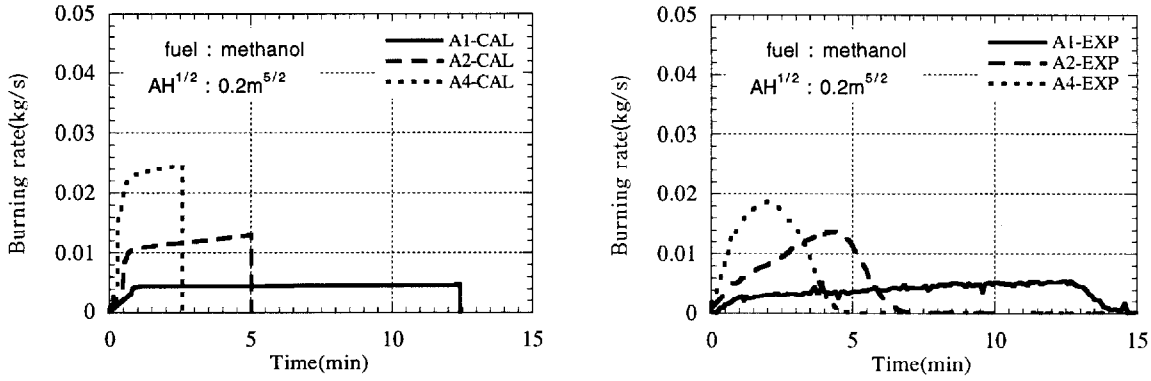

Figure 5. Mass burning rate

Figures.6 -8 compares the predicted and the measured burning rates at vigorous burning stages. The burning rates of the predictions were taken from the values at the last moment of the fire duration, and the value of the experiments are the average during the vigorous burning stage. Again, the open symbols indicate the cases with external flames and the solid symbols indicate the cases without external flames. The solid line in each figure stands for Eqn.(2), i.e. the relationship by Kawagoe et al. and the broken line stands for the rate of fuel which can be burned within the fire room by stoichiometric combustion with the air supplied by the ventilation through the opening,

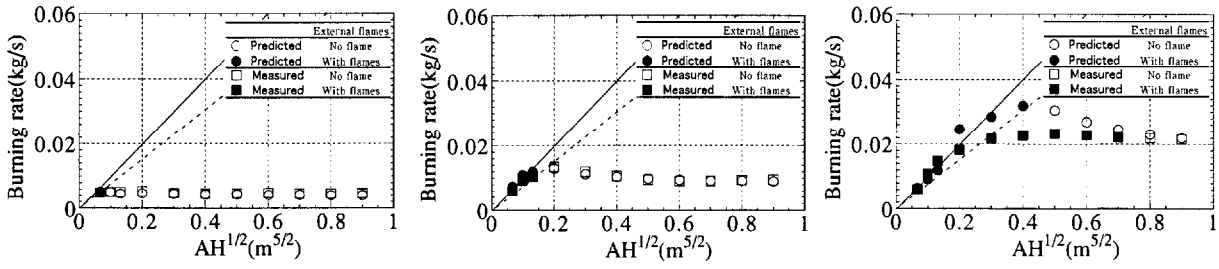

Figure 6. Comparison the predicted and the measured burning rates (Methanol)

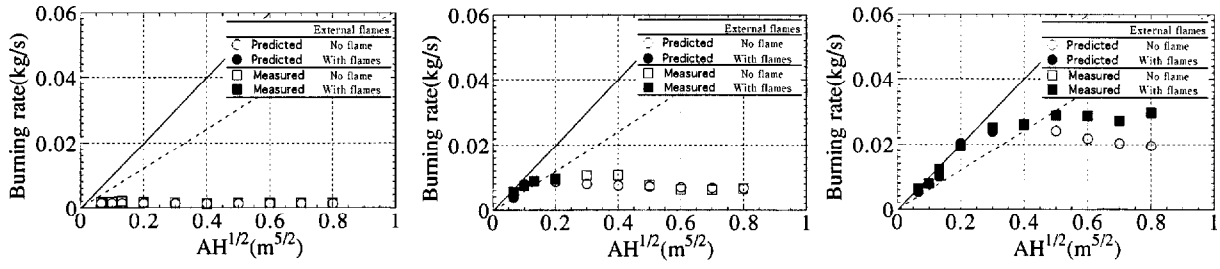

Figure 7. Comparison the predicted and the measured burning rates (PMMA)

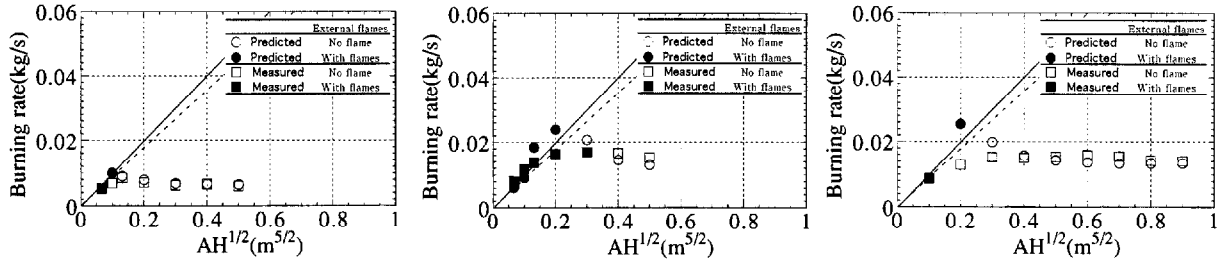

Figure 8. Comparison the predicted and the measured burning rates (Wood)

i.e:

$$M_{stoi} = M_{air} / \gamma \qquad \cdots (27)$$

In this particular case, the stoichiometric burning rate of each fuel M_{stoi} is given as

$$M_{stoi} = \begin{cases} 0.077A\sqrt{H} & (methanol) \\ 0.061A\sqrt{H} & (PMMA) \\ 0.089A\sqrt{H} & (wood) \end{cases} \qquad \cdots (28)$$

where the ventilational air supply rate is estimated by

$$M_{air} = 0.5A\sqrt{H} \qquad \cdots (29)$$

Generally, both of the predicted and measured burning rates tend to increase along the line of Kawagoe et al. regardless the fuel type where the ventilation factor is small and decrease down to the respective value of free burning of each fuel as the ventilation factor becomes large. It seems the maximum burning rates often appear on the broken line, i.e. the stoichiometric burning rate line.

In a couple of the figures, some discrepancies are observed between the predicted and measured burning rates, particularly around the peak burning rates. Partly, this should be attributed to the crudeness of the model, but it should be remained that the measured values are average while the predicted values are the maximum. Considering the complexity of the phenomena involved and the simpleness of the model, the discrepancy to this extent may be acceptable.

5. CONCLUSION

In this study, a room fire model incorporating the prediction of the mass burning rate was developed. The room temperature and the mass burning rate predicted by the model were compared with the results of the preceding reduced scale room fire experiment.

The model seem to agree reasonably well with the test results, despite of its simplicity. The relationship on the mass burning rate of ventilation controlled room fires empirically obtained by Kawagoe et al. for wood crib fire was reproduced by the model regardless the fuel type.

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NOMENCLATURE

A	: Area of opening (m^2)
A_{fuel}	: Area of fuel (m^2)
A_{wall}	: Area of wall (m^2)
c_P	: Thermal capacity at constant pressure(kJ/kg/K)
H	: Height of opening(m)
$\triangle H_{air}$	: Heat of combustion per unit mass of air consumed(kJ/kg)
$\triangle H_{fuel}$	: Heat of combustion per unit mass of fuel consumed(kJ/kg)
k	: Effective absorption coefficient(1/m)
L	: Thickness(m)
M	: Mass burning rate(kg/s)
M_{stoi}	: Stoichiometric burning rate(kg/s)
m''_{free}	: Mass burning rate per unit surface area in the free(kg/s)
P	: Partial pressure(atm)
Q_C	: Rate of heat release by combustion (kW)
Q_{fuel}	: Rate of the heat transfer to fuel(kW)
Q_{ex}	: Heat release rate due to the combustion of excess fuel(kW)
Q_R	: Radiative heat loss from the opening(kW)
Q_S	: Heat convected out by the opening flow(kW)
Q_W	: Heat transfer to the boundary walls(kW)
L_V	: Latent heat of gasification(kJ/kg)
R_{OF}	: Rate of air inflow(kg/s)
R_{FO}	: Rate of air outflow(kg/s)
T	: Fire room temperature(K)
T_O	: Outside (ambient) temperature (K)
T_P	: Fuel gasification temperature(K)
T_W	: Surface temperature of the wall(K)
V	: Volume(m^3)
$Y_{1\cdot O}$	: Species mass concentration(outside)
Y_1	: Species mass concentration(inside)
h	: Convective heat transfer coefficient($kW/m^2/K$)
γ	: Stoichiometric air/fuel ratio
Γ_1	: Species generation(consumption) rate(kg/s)
ε_G	: Emissivity of room fire gases
ε_W	: Emissivity of wall surface
ρ	: Density of air(kg/m^3)
σ	: Stefan-Boltzmann constant($=5.67\times10^{-11} kW/m^2/K^4$)

Discussion

John Rockett: One of the attractive features of the simple models is the possibility that they can be programmed using software such as Mathmatica. One can arrive at solutions with little programming effort. I commend you on your model and I will have a look to see if I can make it run with Mathmatica.

As an old modeler, it would be interesting to compare the results of your model with what you would get with running the Harvard single room fire model.